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Static and Dynamic Analysis of Suspension System with Longitudinal Spring and Damper Assembly

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***Abstract*— Suspension systems have changed rapidly over past six or seven decades. Most of the designs come with the passive suspension system which gives freedom to the designers to accumulate most of the components in a compact space. This system will give designers the benefit of accumulating the suspension ahead of the wheel which in fact will be helpful in reducing the un-sprung weight on the front wheels compared to 'A' arms suspension system. Moreover it would be generously helpful to analyze such kind of suspension system to study various factors as spring force, space requirements and road handling characteristics.**

***Index terms*—** Suspension System, Dynamic abilities of Horizontal Spring and Damper assembly.

I. Introduction

Suspension systems have been changing and characteristics like roll handling, Drivability, Suspension predictability have been better and better. Moreover it is the only part which connects the sprung and un-sprung mass of the vehicle together.

The other purpose of the suspension system is to isolate the ride from road irregularities, vibrations and provide a comfortable ride. Moreover a good suspension system takes up stress out of various important components like the knuckle and also helps reducing steering effort. Also the suspension system has important characteristics of road holding and reduces the drive line vibrations ensuring a comfortable ride for the passengers. A good suspension will always help to connect the wheels to the ground.

In all suspension systems that have come up, the main function of the suspension system is to reduce strain from various components. When a vehicle goes over a bump the strain from all the components is stored as spring force in the spring which in turns is released back as soon as the external force on the wheel is taken off. In vehicle dynamics terms we call it as Jounce and Rebound. The vertical vibrations caused by the suspension system needs to be absorbed for which the damper is placed along with the spring.

This suspension system was designed to study various factors involved in suspension design and to study the loading conditions for such a system.

II. Suspension System setup and design considerations

The system consisted of the swing arm and the spring and damper assembly at the front considering the total weight of the vehicle and the space constraints. Then a geometric model was created in Pro-E and was simulated in ANSYS and later in ADAMS for both the static and dynamic load considerations.

A. Design Considerations of spring

The load on the front of the vehicle was calculated and hence various springs were detained and used to find a perfect balance between the spring stress and vehicle roll moment. The first thought was taking a mono suspension into account but the stiffness of the mono suspension was way over requirements. Hence we decided to go for double spring and damper assembly and fabricate spring along wise.

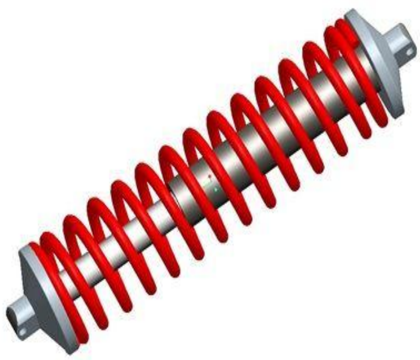

Figure 1: Coil Spring

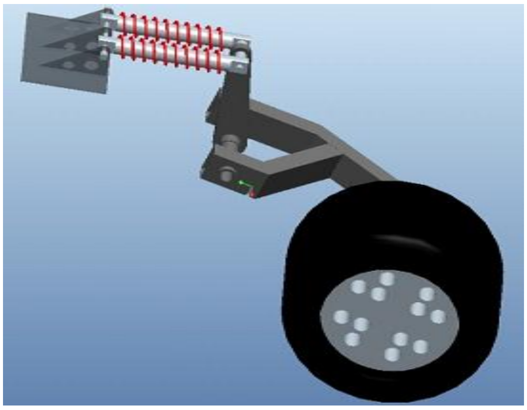

Figure 2: Assembly of suspension system

Spring Stress Calculation

$$\tau = \frac{8WDK_w}{\pi d^3}$$

Where
D, Mean Coil Dia = 57 mm
d, Wire diameter = 8 mm
K_w, Wahl's Correction factor=1.278
W, Load of the vehicle = 1650N
τ = 597.8 N/ mm^2
Spring Stiffness,

$$k = \frac{Gd^4}{8D^3 n}$$

Where,
N, No. of Active Coils= 10
G, Modulus of Rigidity= 126 × 10^3 MPa
k= 35 N/mm

B. Modelling of swing arm

The swing arm was modeled considering all the space prerequisites and geometric limitations. The leading arm is essentially a work of box type structural steel fabrication model. The leading arm considered was designed under the consideration of reduction of the overall weight of the vehicle and so was so fabricated to have less weight and more strength as compare to the solid members if done so.

The bell crank lever type arrangement was considered to help the swinging action of the leading arm. The bell crank lever gives the option of the transfer of the force easily to the springs and transfer of the force in from a linear motion to a rotational motion. The shorter arm of the bell crank is fitted with shock absorbers on the vehicle frame.

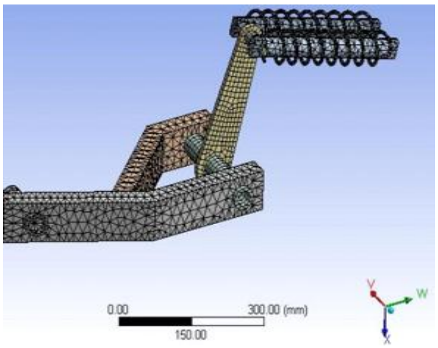

Figure 3: Meshing of suspension system in Ansys

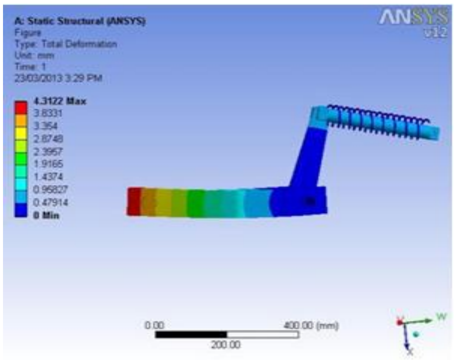

Figure 4: Total deformation of suspension system

The leverage can be varied by changing the length of the bell crank lever depending upon the load requirements. The modeling of the suspension was carried out in Pro-E while the simulations were carried out in ANSYS and MSC ADAMS.

TABLE I
BOUNDARY CONDITION AND MATERIAL PROPERTIES OF SWING ARM AND ITS MOMENT AT VARIOUS POINTS

Particulars	Point 1	Point 2	Point 3	Point 4	Point 5
Volume mm^3	5.6674 e+5	3.6799 e+5	16398	3.1508 e+5	2.1971 e+5
Mass kg	4.4546	2.8924	0.12889	2.4765	1.7269
Centroid X mm	50.148	66.1	102.99	155.11	105.11
Centroid Y mm	-54.525	432.7	281.67	84.907	200.0
Centroid Z mm	-1.0739 e-6	-206.37	-246.84	6569 e-9	-6.8787 e-15
Moment of Inertia Ip1 $kg.mm^2$	1.2211 e+5	25935	3.1487	25933	598.3
Moment of Inertia Ip2 $kg.mm^2$	6728.5	504.17	194.82	3392.6	10670
Moment of Inertia Ip3 $kg.mm^2$	1.2239 e+5	25912	194.82	25264	10670
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Point 1, Point 2-Suspension Hinged Point
Point 3, Point 4-Welded Points on Suspension
Point 5- Point Where Wheel is Attached

C. Analysis of swing arm under static loads

Figure 5 shows analysis of swing arm in the direction of travel of the wheel under a bump. The support is kept fixed at one side while a force of 14700N is applied near the sides of the wheel. When load is applied at one side of the swing arm the max deformation under load was 4.302mm which is considerable considering the length of swing arm which is more than 410mm.

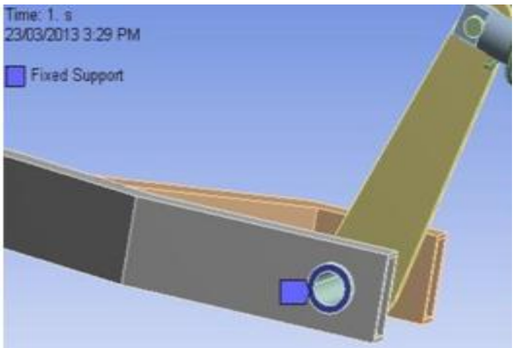

Figure 5: Fixed Support

Figure 6 shows equivalent stress under the applied load which is 609.96 Mpa. The maximum equivalent stress is more on the angular contacts rather than on the beam structure.

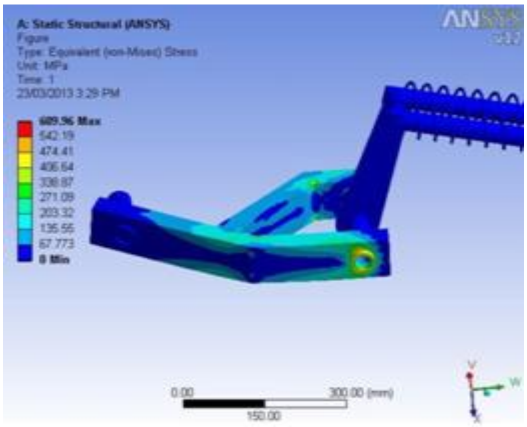

Figure 6: Equivalent (Von mises) stress

Figure 7 shows the Maximum Principal Stress applied on the applied load which is 592.57 Mpa. The maximum principal stress is on the fixed point, corners and the contact points of the swing arm.

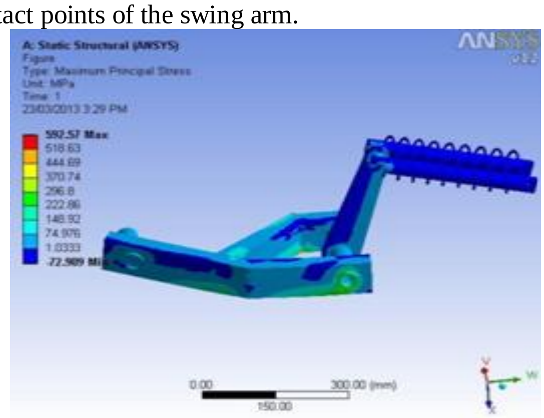

Figure 7: Maximum principal stress

D. Dynamic Analysis of Suspension Systems

The dynamic analysis was carried out in MSC AD-MAS, a multi-body simulation software has consid-ered all the constraints applied on the suspension sys-tem. Fixed points were given to hold the suspension during the working.

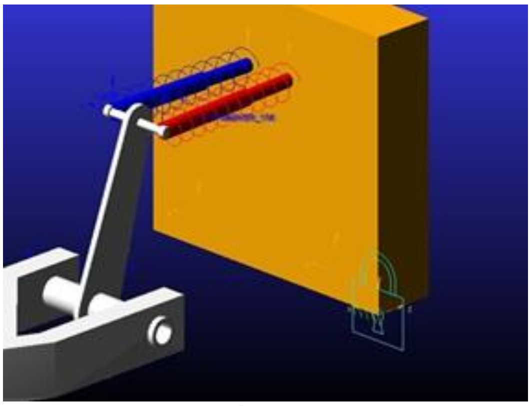

Figure 8: Fixed support in ADAMS

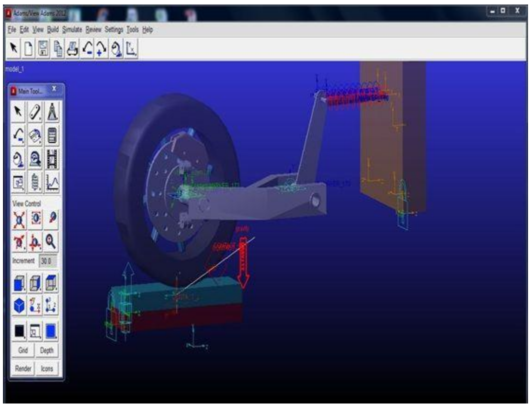

Figure 9: Direction of Travel

The fixed points on the vehicle hold the quarter car model to hold in place so that the forces and analysis results can be obtained. The size of the tyre contact patch has been fixed and deflection of suspension under various bump travels has been defined.

TABLE II: DEFLECTION

Deflection	Description
50 mm	Rambling strip (small continuous bump), height difference while transferring from one to another road surface
100 mm	Normal bumps
150 mm	Heightened bumps (Speed breakers)

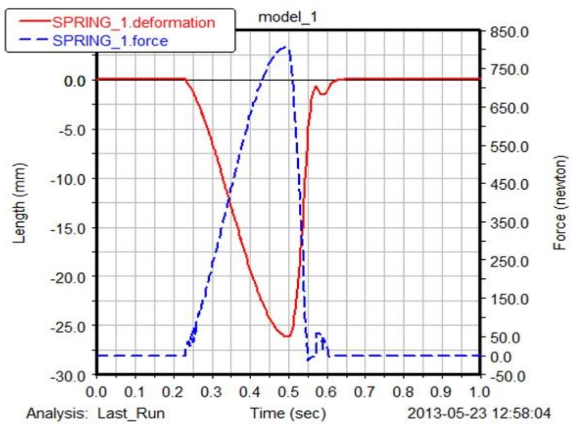

Figure 10: Spring1 Deformation Under 800N load

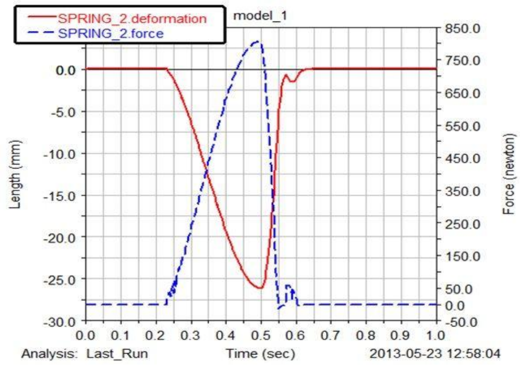

Figure 11: Spring2 Deformation under 800N load

Figure 10 & 11 shows spring 1 and spring 2 deformations under application of load of 800N. The deflection is 27mm on the spring for 50mm of bump travel.

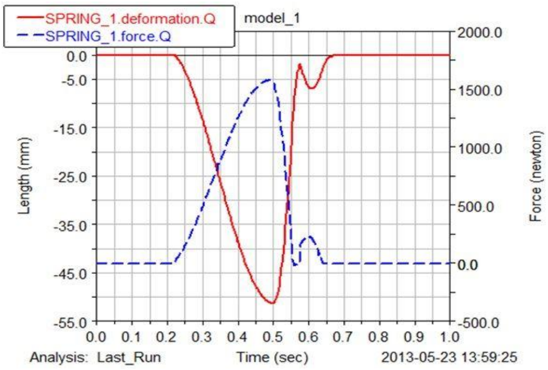

Figure 12: Spring1 deformation under 1650N

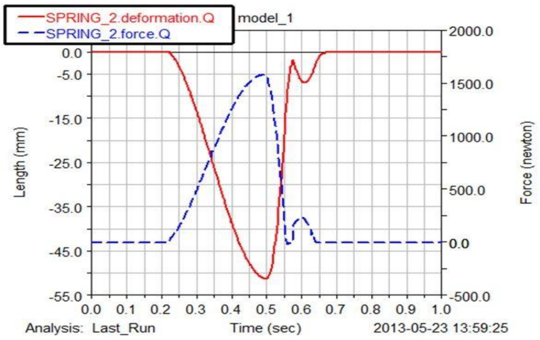

Figure 13: Spring2 deformation under 1650N

The above two figure shows deformation of spring under 100mm bump travel is 1650N while the deflection recorded on the spring was 52mm.

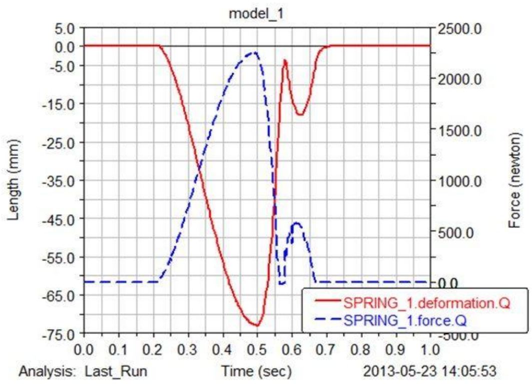

Figure 14: Spring1 deformation under 2250N

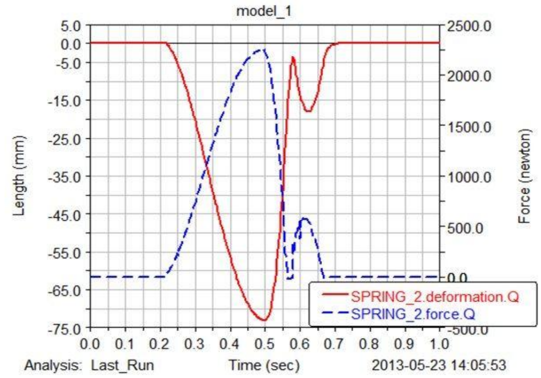

Figure 15: Spring2 deformation under 2250N

While the force under 150mm bump travel is 2250N, the deflection on the spring was 74mm.

TABLE III
FORCE LONGEVITY

Force Longevity (How much time does the force last on dampers) (s)	Spring movement (mm)	Applied Load (N)	Bump Dimension (mm)	Time Period of Application (s) (start,Max deflection, end of deformation
0.4	27	800	50	(0.2, 0.35, 0.65)
0.2	52	1650	100	(0.2, 0.3, 0.62)
0.6	74	2250	150	(0.2, 0.38, 0.63)

III. CONCLUSION

The leading arm was given the steering angle compensation and the leading arm model was modeled in the software Pro-e. The inclined model has given the suspension system a good space freedom to allow the steering system location and better handling and good work space. The model analysis shows that the model works better than the previously considered model with straighter arm.

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